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## SOLAR WATER HEATER TEST PROJECT



*Prepared for*

***MONTANA DEPARTMENT of NATURAL RESOURCES and CONSERVATION***

# SOLAR WATER HEATER TEST PROJECT

Prepared by

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Insolation Measurement
Solar Performance Monitoring
Solar/Conservation Design
Instrumentation/Testing
Data Acquisition/Processing

FINAL REPORT, GRANT AGREEMENT RAE-83-1029

SOLAR WATER HEATER TEST PROJECT

OVERVIEW

The tests outlined in the grant agreement have been completed. Test data for a one-year period have been collected on three solar water heating systems and one electric water heater.

This report has been organized into one section which discusses the overall results and recommendations and a final technical section discussing the engineering analysis leading to the results.

The primary data was the heat output of all the systems and the corresponding environmental data during each hour for one year. This was summarized into monthly totals and averages and annual total energy outputs. The active solar system provided the most solar heat followed closely by the intergral passive system. The homemade system preheated the water during the summer.

Assuming electricity at $0.04/KWh the electric water heater (80% efficiency) would have used $243 worth of electricity for the test year. This matches almost exactly the cost on the yellow "energy label" on the unit.

Using the same assumptions the active solar system saved $136, the IPH system saved $95 and the homemade batch system saved $26.

The installed costs of the active and IPH system are estimated at $3500 each and the cost of the batch system about $800. These installed costs could vary considerable depending on the site.

The IPH and batch systems can freeze during cold weather which is a practical consideration. The IPH system can produce dangerously high water temperatures and should be used with a tempering valve. The supply and return plumbing of the passive systems has a significant effect on their efficiency and thermal performance. These aspects are addressed in detail in this report.

CONCLUSION

These tests have provided data which will be useful to the Department of Natural Resources, Montana Power Company, and consumers in evaluating this solar application.

Parts of this data were used by BPA, Oregon Department of Energy and University of Wisconsin Solar Energy Laboratory for technical evaluation of passive system performance. A summary of this data is scheduled for an article in Solar Age magazine.


Figure 1: Solar water heater test site

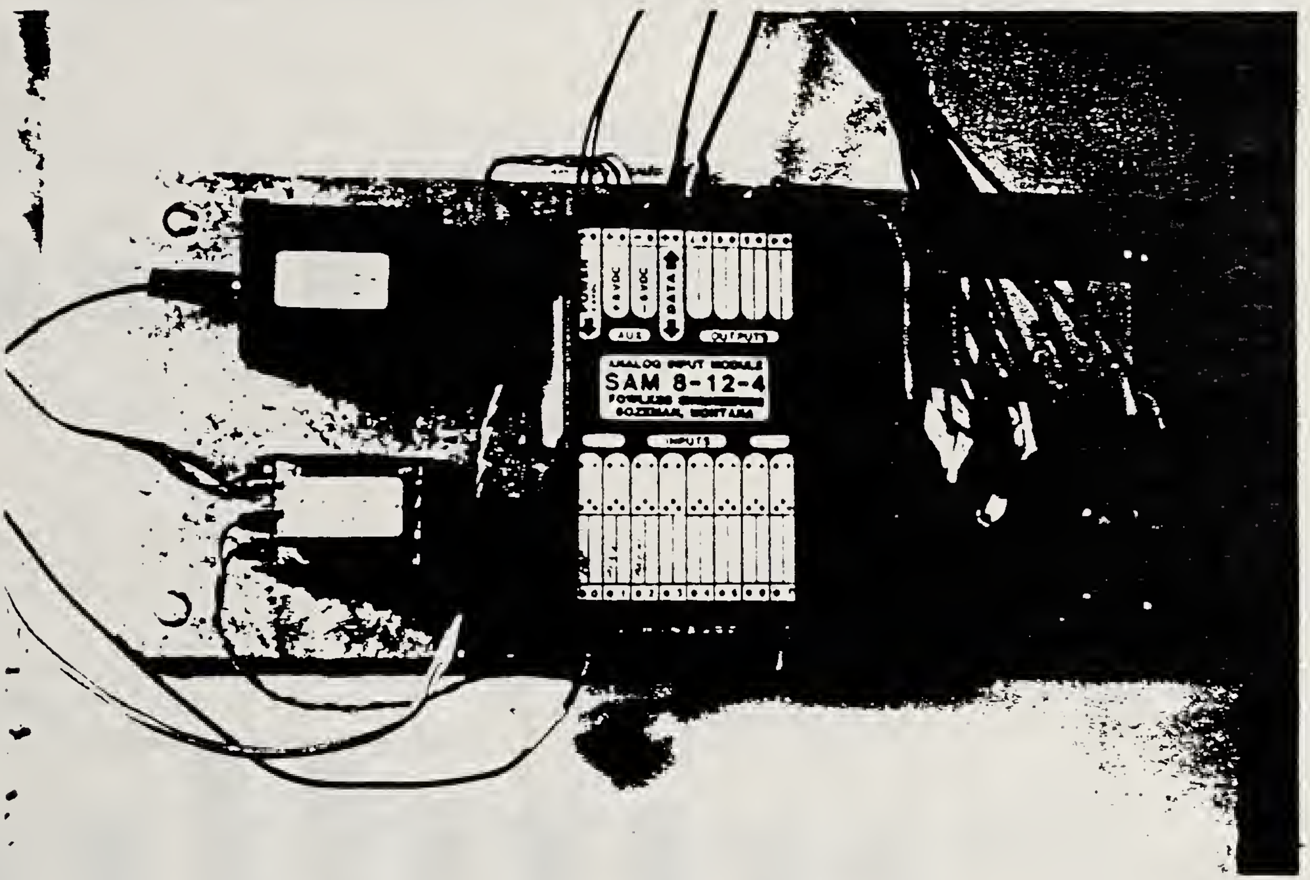


Figure 2 Data acquisition system

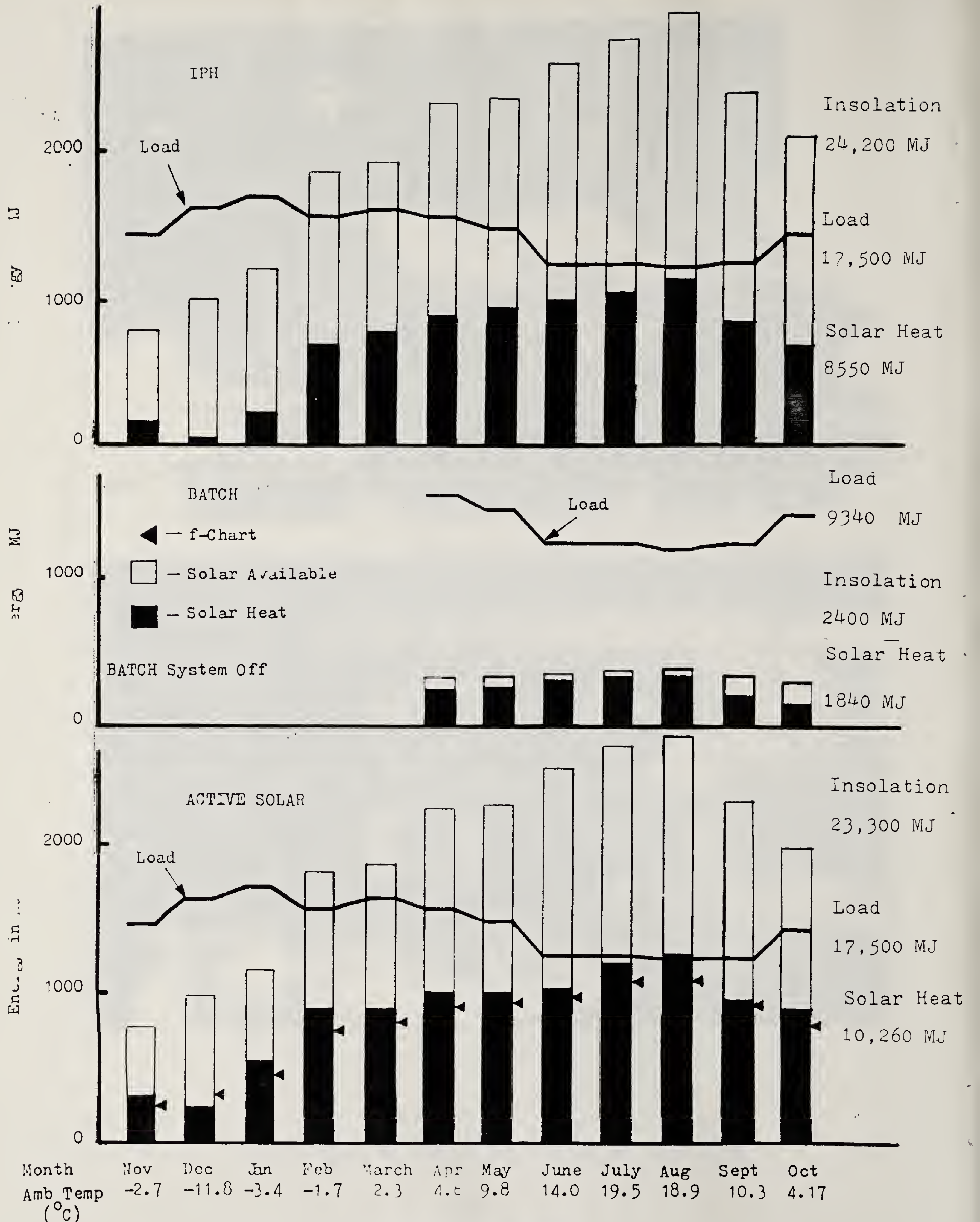


FIG 3:

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SECTION 1: SUMMARY OF PERFORMANCE DATA
(submitted to Solar Age Mag.)

DEEP FREEZE TEST OF SOLAR WATER HEATERS

The thermal performance of one active and two passive solar water heater systems were carefully monitored during the past year in Bozeman Montana. Record-setting cold temperatures created a unique test environment for the passive solar system. The computer based data acquisition system tracked the thermal response of the systems through periods of extreme low temperatures. This data showed a remarkable fact:

1) The tanks in the passive heaters did not burst.

ANATOMY OF A FREEZE-UP

Figures 1 and 2 show graphs of tank temperature, ambient temperature, connecting pipe temperature and solar radiation during a cold period beginning on December 17.

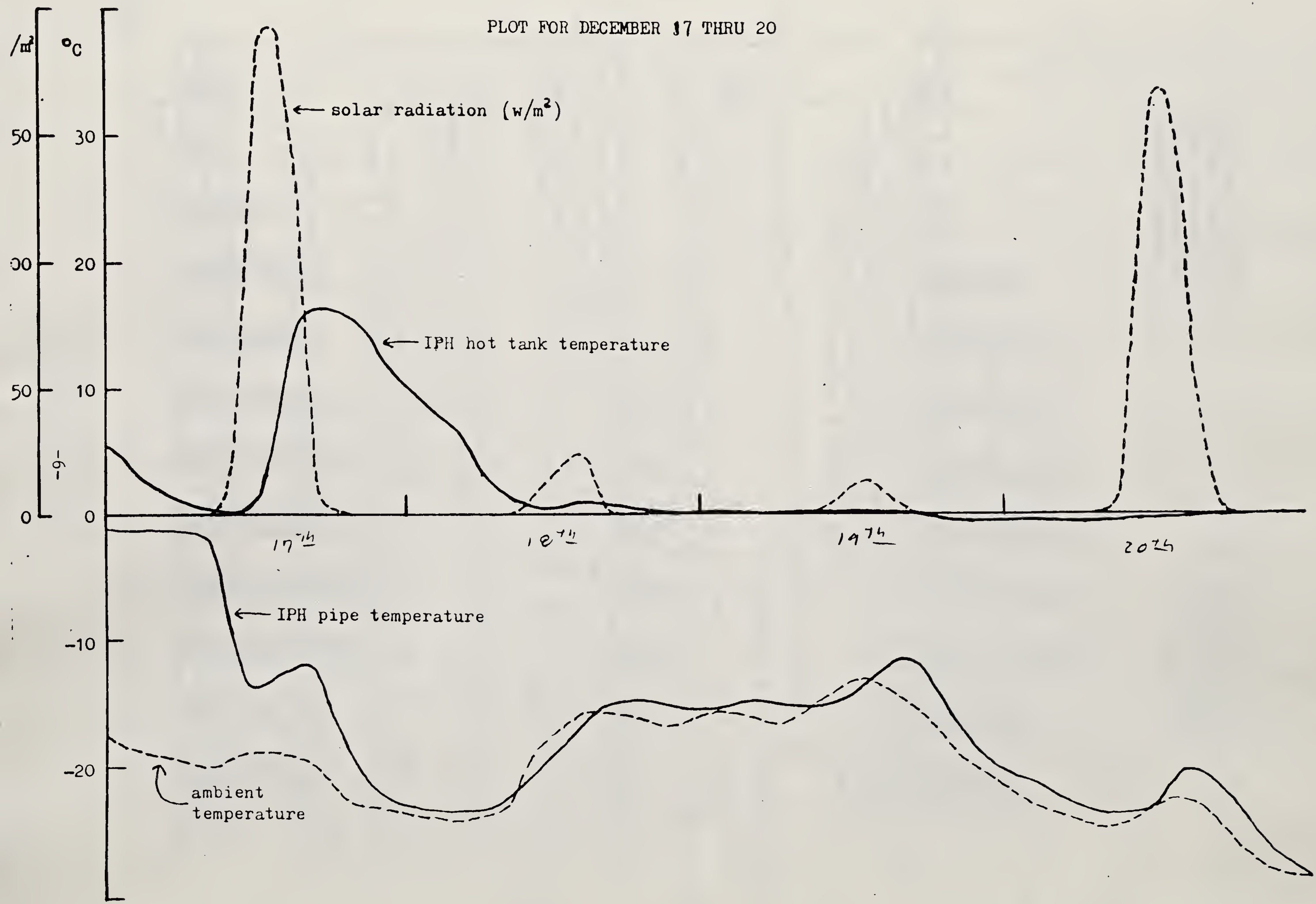
[a] Just before sunrise on Dec 17 the outside pipe froze. For the rest of the graphed period the pipe temperature pretty much followed ambient temperature with a few departures due to the sun heating the outside of the pipe. Due to the frozen pipe there was NO FLOW so that the passive solar collector is "stagnated" for the graphed period.

[b] The temperature of the stagnated tank reflects the solar heat gain during the sunny day on Dec. 17 and the subsequent losses at night.

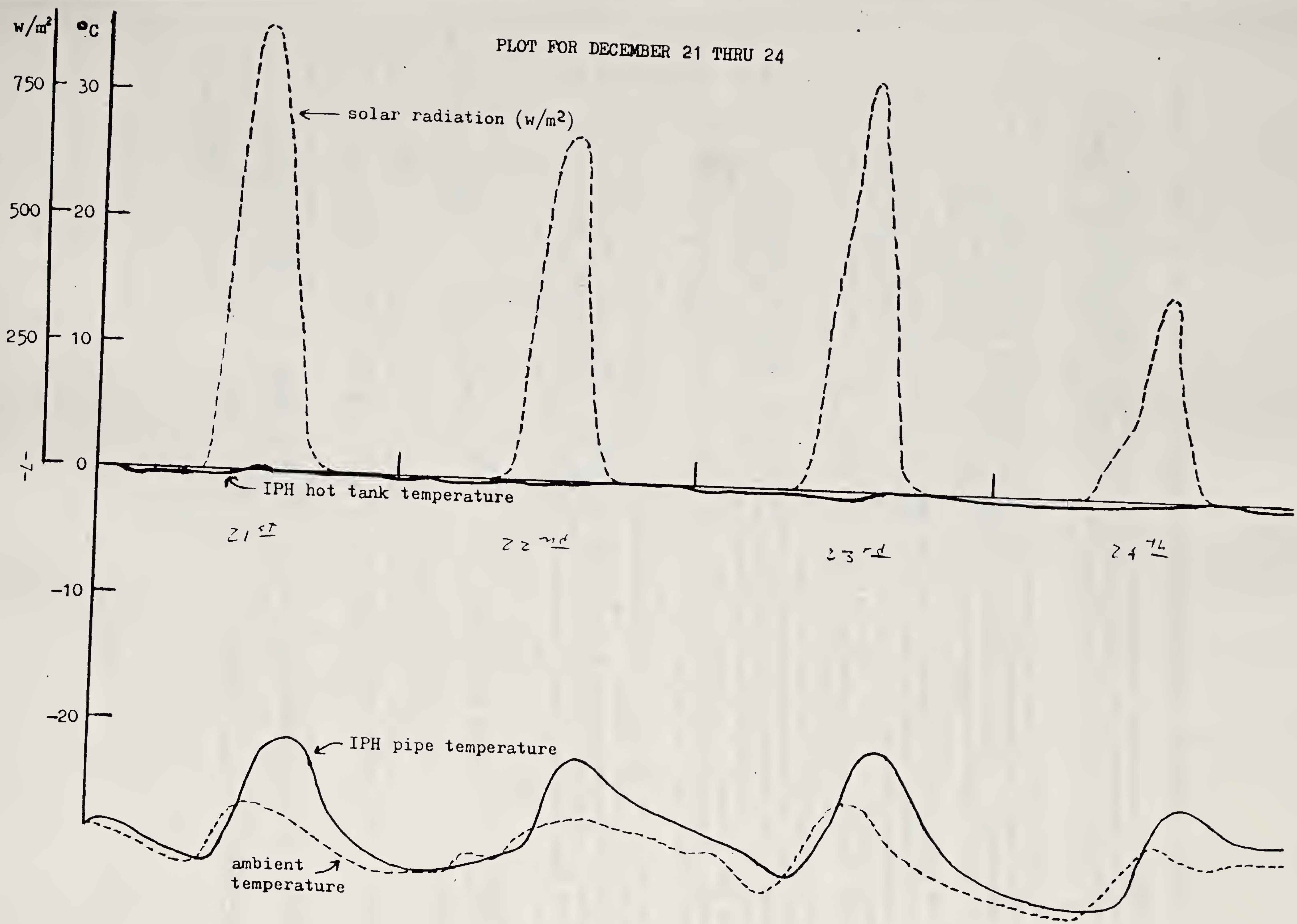
[c] Dec. 18 and 19 were overcast with ambient temperatures rising to about -15 degC! The tank temperature is 0 degC indicating that the water is partially frozen.

[d] The skies cleared on the night of Dec 19. Ambient temperatures dropped below -20 degC. The next five days were clear and very cold.

[e] During the five cold/clear days the tank temperature remains at 0 degC through day and night. This implies that the water was always some mixture of liquid and solid; during the day the sun would melt some of the ice which formed again the next night.


PLOT FOR DECEMBER 17 THRU 20
/m²
°C
50
30
00
20
50
10
0
0
-10
-20
solar radiation (w/m²)
IPH hot tank temperature
IPH pipe temperature
ambient temperature
17th
18th
19th
20th

PLOT FOR DECEMBER 21 THRU 24



## ACTIVE, PASSIVE, HOMEMADE DESCRIPTIONS

Table 1 describes the three solar systems at our test site. The active and passive systems had similar solar aperatures and tank sizes so it is valid to compare them directly. Their cost also is similar.

We believe the particular active system we chose for this experiment is representative of active systems available to consumers. This particular system contains quality components, is well integrated and widely marketed at an attractive price.

The manufactured passive system uses double glazing and incorporates a selective surface absorber. It has an ample aperature and the back of the collector is well insulated (R22). We judged that this combination of design features would be the best choice for operation in our cold climate.

The hommade system was included in the test to represent a minimal, do-it-yourself effort. We believe that there may be many other homemade designs that would have better performance than the one we built. The aperature of our design, for example, is quite small. It seems clear that most do-it-yourselfers could easily immitate many of the more sophisticated commercial designs.

One half of the metal enclosure was cut from a standard, low cost electric water heater. The face of the tank was painted with black paint and glass fiber insulation was stuffed into any holes. A single layer of Filon was pop-rivited where the metal case was removed. Silicone was used to caulk all joints.

| TYPE | MODEL / MFG | APERATURE $m^2$ | TANK CAP, GAL |
|---|---|---|---|
| ACTIVE | SOLAR CRAFT / STATE | 4.32 | 80 |
| IPH | SUNFLAME / ENTECH | 4.49 | 78 |
| BATCH | HOMEMADE | 0.62 | 45 |
| ELECTRIC | STATE | ∅ | 50 |

TABLE 1 SYSTEMS

## ACTIVE, PASSIVE, HOMEMADE PERFORMANCE COMPARISONS

Figure 3 shows the thermal performance summaries of the three solar systems for ten months. The bars show how much solar energy was intercepted by the collecter and how much solar energy was delivered to the basement as heated water.

The LOAD curve shown in the graph is the amount of hot water delivered by an electric water heater set at 50 def C (120 degF). (This heater was run as a control for the solar systems. The electric heater had R16 insulation and had an overall efficiency of 80%). The variation in the load curve is due to annual variations in cold water supply temperature. The water was drawn hourly according to the Rand distribution giving a total draw of about 73 gallons per day.

The active and manufactured passive system's outputs are similar. Over the ten month period the active system produced 7552 MJ (after deducting pumping energy) and the passive system produced 7015 MJ. An f-chart prediction was generally conservative by several percent.The active system is thus the winner by a small margin when comparing either monthly or annual heat output.

During any sunny day, the passive collector is more efficient than the active collecter,i.e. the water in the passive tanks is hotter. This is because the sun heats the water in the tank directly instead of through two heat exchangers.

During the night, however, the active system storage tank is in the house while the passive storage tank is on the roof. To make things worse, the passive tank is not insulated as well since one -half of its enclosure consists of glazing. These night-time losses pose a considerable penalty on the average performance of the passive system; it may have been hotter at the end of the day but by the next morning it is colder than the active system storage tank.

Because of the importance of night-time heat losses the performance of the passive system will depend more strongly on the daily load curve. The passive system has a "use it or loose it" characteristic that cannot be entirely avoided.

The Rand curve used in this experiment has relatively more water use in the evening. If large draws were made in the morning the passive system would have been less effective. On the other hand, if solar heated water had been used every evening following a sunny day, the overall system efficiency would have been significantly higher.

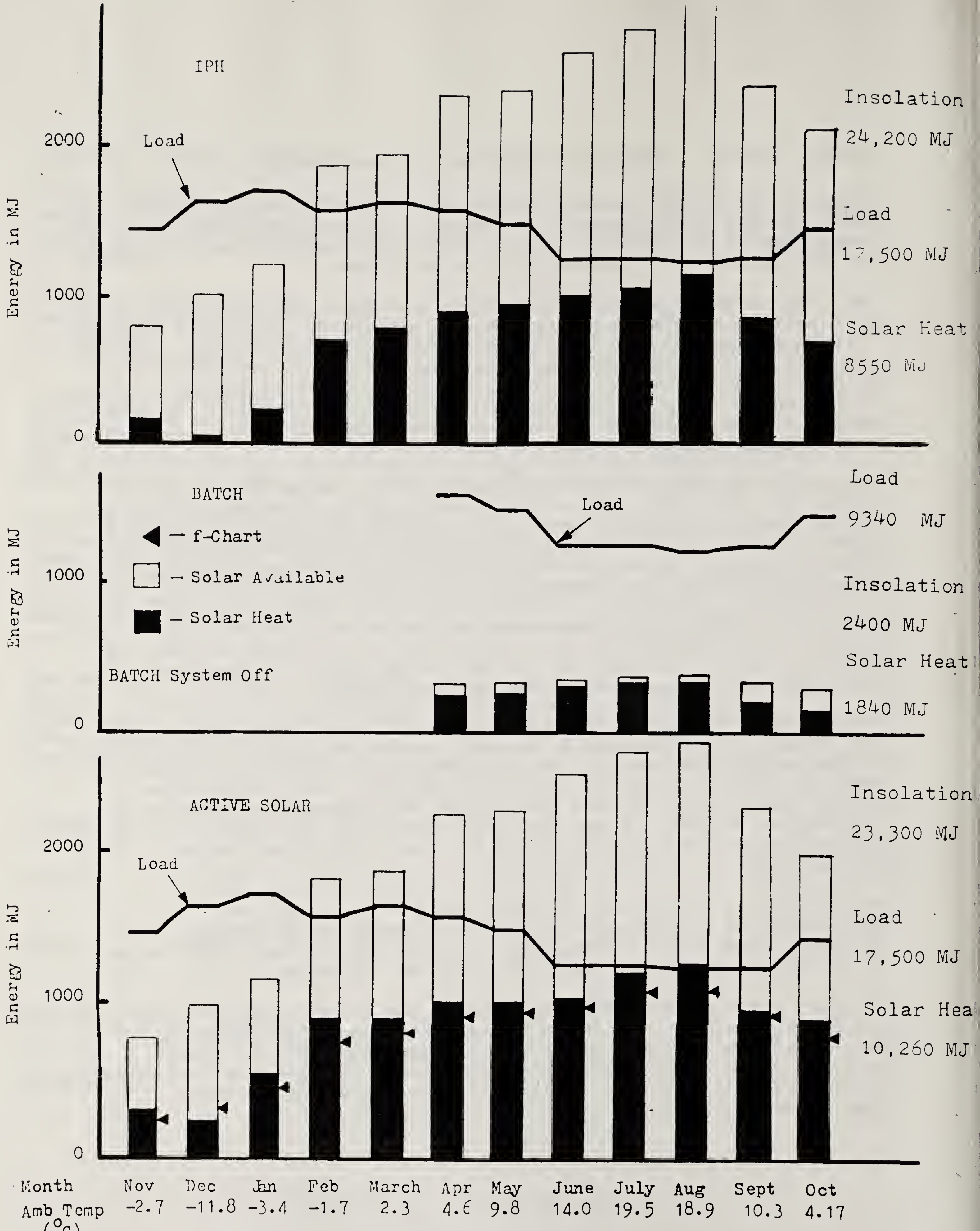



FIG 3:

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## EFFECT OF SUPPLY AND RETURN LINES ON SOLAR HEAT OUTPUT

The passive water heater is almost always mounted some distance from the point of use of the heated water. This inherent geometry gives rise to some unique thermal effects.

During cold weather any supply or return lines that are outside the building will tend to freeze long before the tanks begin to freeze. This is because the lines have more surface area per unit thermal mass than the tanks.

Once the lines are frozen the collector is "stagnated" and any solar radiation absorbed will help protect the tank from freezing; no solar heat goes to the load. This sequence forms a rather interesting "self preservation" mechanism for the tank. The frozen solar heater must of course be bypassed to service the auxiliary heater during this period.

In the Montana experiment the outside supply lines were plumbed with polybutylene pipe with R 7 insulation. During the winter these lines went through several freeze-thaw cycles. No leaks were observed.

The supply and return lines affect the system's thermal performance during normal operation. In our test there were about 15 feet (each) of supply and return pipe inside the heated space insulated to R3. Outside was about 12 feet of pipe isulated to R7. This amount of piping would be fairly typical of most residential installations.

Between draws the cold water in the supply line absorbes heat from the building. The exterior portions of the cold supply and "hot" return also exchange heat with the ambient air. The "hot" return line exchanges heat with the building between draws.

We wanted to evaluate the effect of these piping gains and losses in our experiment. A computer model of the piping was constructed. The piping was divided into elements and each element was assumed to experience Newtonian heating (or cooling). Hourly environmental and system temperatures were input to the model and the net piping heat gain (or loss) was computed each hour. These hourly values were summarized into monthly values.

During the winter months there was a net GAIN of heat in the piping system. This gain was 10 to 20% of the total "solar" gain measured at the system output. The piping gains became small during the spring and starting in May there was a net heat LOSS from the piping. During July the piping loss was about 5% of the total solar heat delivered by the system.

Heat exchanges in the supply and return piping are thus seen to be relatively small as compared to the annual solar system heat output. The winter pipe gains about balance the summer pipe losses. Unfortunately during the winter the system is stealing heat from the house, when it is needed, and returning it during the summer, when it is not needed.

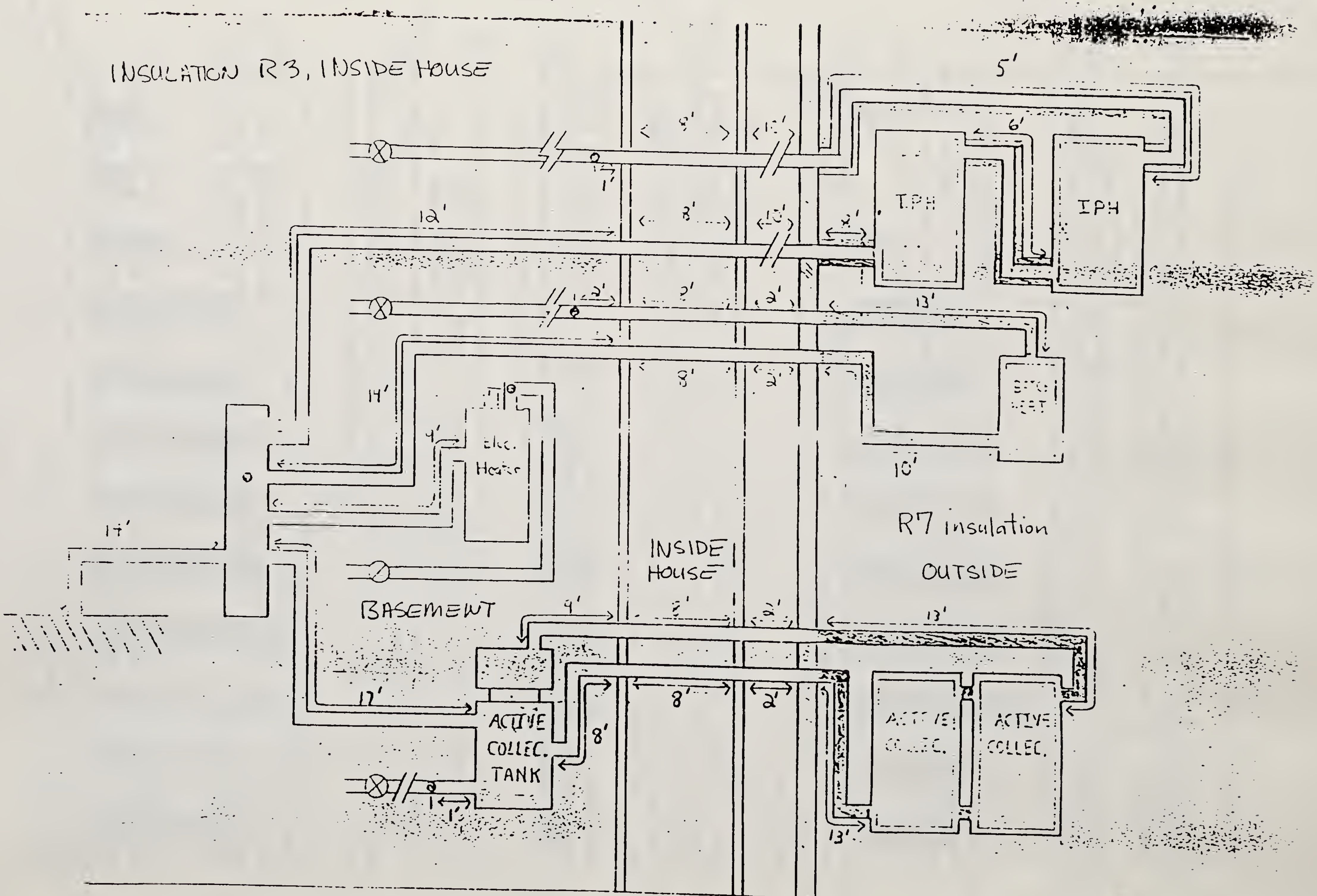


Fig. 4: Pipe Lengths and Insulation

This sensitivity to load timing should be kept in mind by anyone attempting to interpret field data on passive solar water heaters. One should expect considerable scatter in field data for solar fraction, etc. due to this characteristic. Policy dicisions must also address this issue when trying to access the impact of this type of solar system.

During the summer the passive system often delivered water that was above 60 degC. We had two test periods this summer when there were no draws for two or more sunny days. The water temperature during the first draw after stagnation was 80 to 100 degC! In real applications, a tempering valve should be used to prevent scalding.

During the winter, the pipes on the passive system froze several times. In a real application this calls for a by-pass cold water supply to prevent interruption of the hot water supply to the house.

The thermal performance of the homemade system has several unique features. The bar graphs show that is meeting only 21% of the load. In view of its small aperature this performance is not bad. The average solar efficiency appears to be 82% which quite high. There are several facts which help explain this efficiency:

a) The evening draws effectively "empty" this system of solar heated water due to its small tank capacity. Since it sits on the roof all night filled with cold water its night-time losses of solar heat are necessarily small.

b) The supply and return piping are net heat gainers which account for about 11% of the average "output".

c) The low operating temperature results in low average heat losses, even during the daytime.

The factors contribute to the high apparent thermal efficiency. The delivery temperature is seldom above 20 or 30 degC. This is too cold to use for a bath; thus the system as presented is strictly a solar pre-heater for some other water heater.

SUMMARY

1) The active system provided more solar heated water during the test period.
C
2) Night-time standby losses penalized the performauce of the passive system.

3) Freezing and over-temperature were operational problems for the passive system within the context of our test. These problems would probably occur in many real applications.

4) The hommade passive heater was low in cost and an effective solar pre-heater.

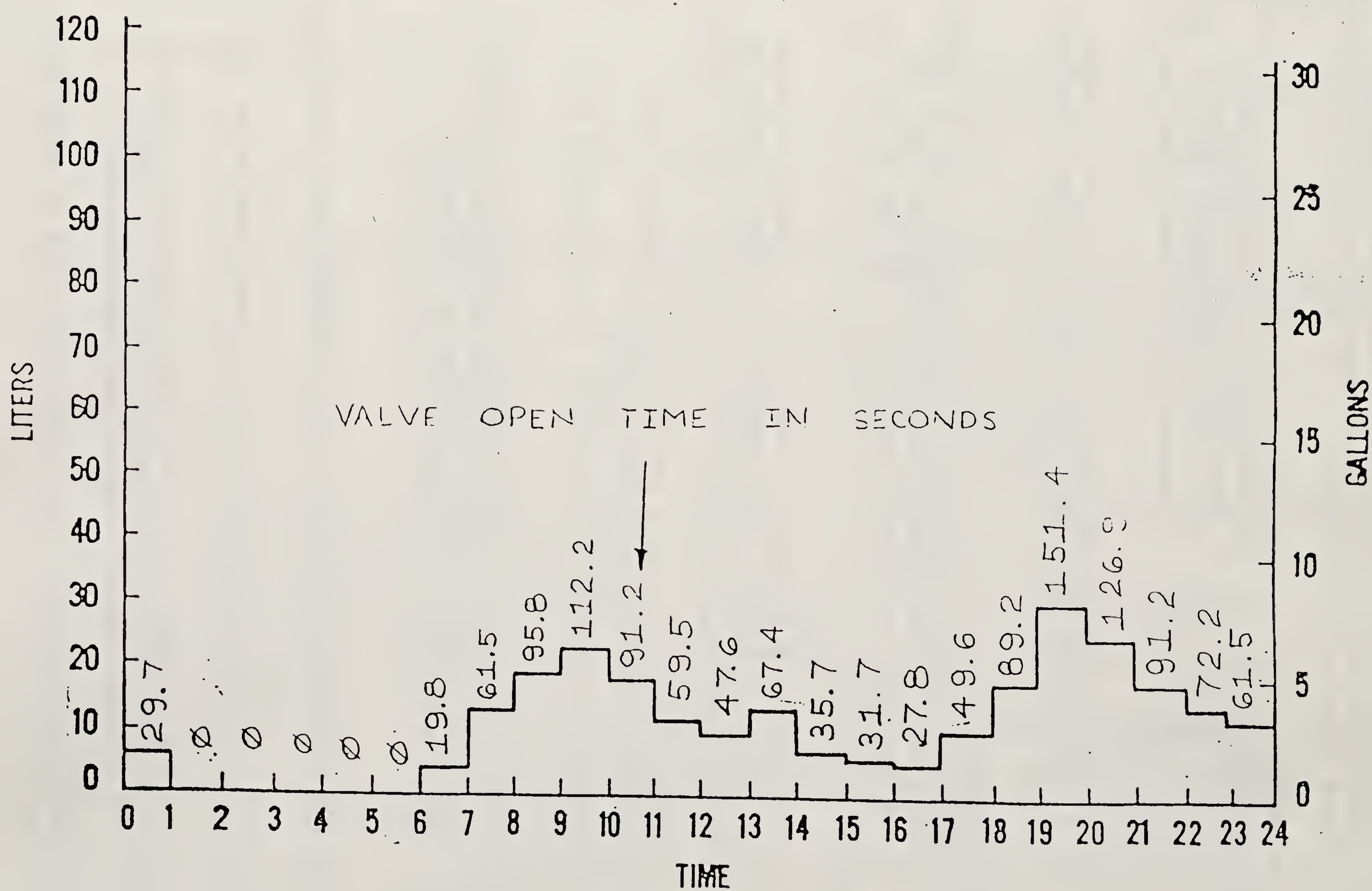


Fig. 1 Rand Hourly Draw Profile

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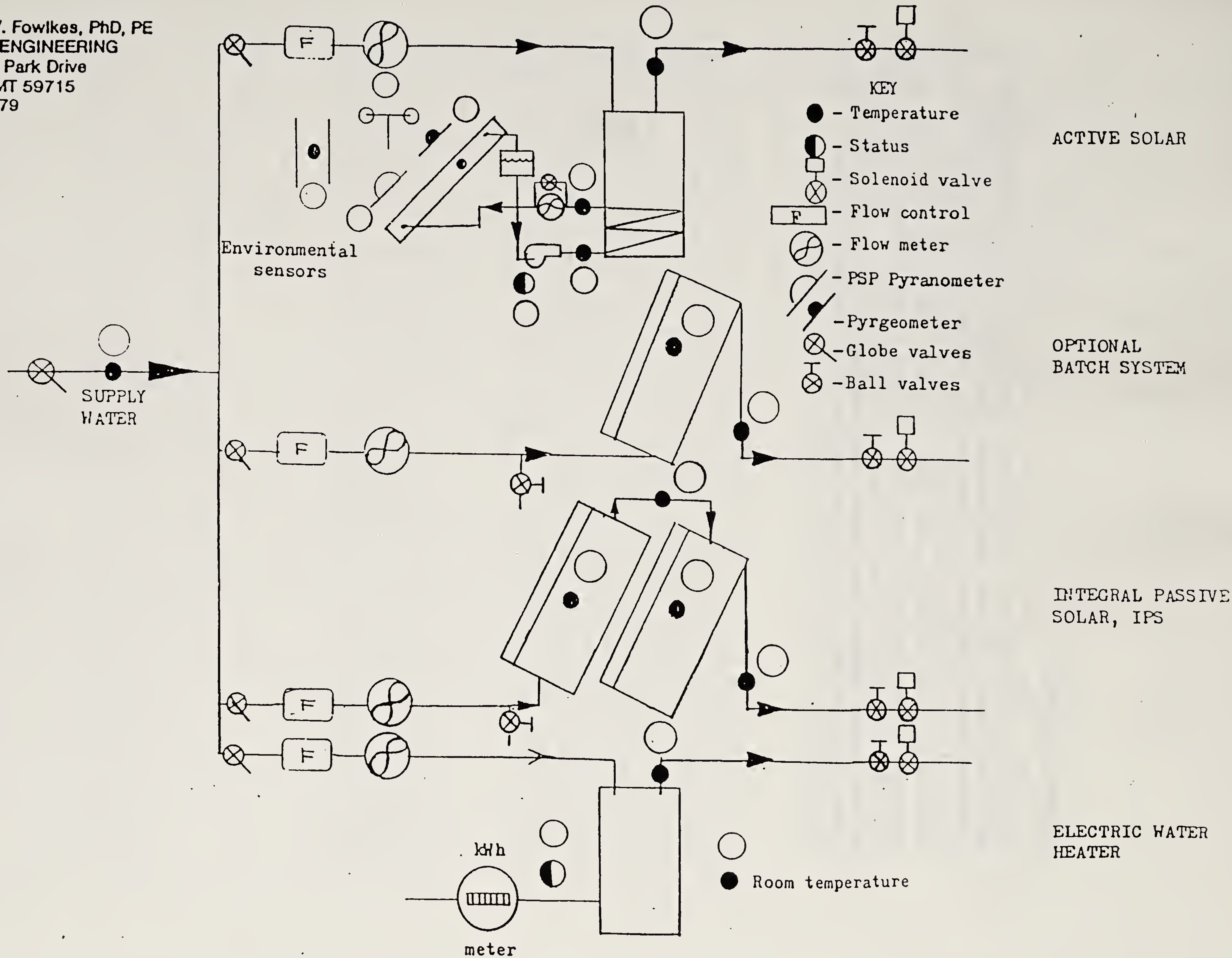


Figure 1: Instrumentation plan and test arrangement, Oct. 27, 1983.

ACKNOWLEDGEMENTS

The Montana Department of Natural Resources and Conservation provided about 90% of the funding for this research project. The Montana Power Company provided about 10% of the support including use of their building for a test site. Bonneville Power Administration participated during the later phases of the test project. The portable computer and SAM 8.12.4 Data Acquisition Module were contributed by Fowlkes Engineering. The Solarcraft active solar system hardware was donated by J. N. Marshall Co. of Denver, Colorado. The participation and support of the organizations listed above are gratefully acknowledged.

## Introduction

The Integral Passive Heat system consists of two water storage tanks and its associated pipe network. The batch solar water system is a passive system consisting of one storage tank and necessary plumbing. Table 1 lists the storage capacities, apperature areas, thermal masses and insulation areas of the IPH and batch solar water heating systems.

## IPH System Heat Losses

The IPH system heat losses consist of two components. These two components are heat losses from the storage tanks and heat losses from the pipes.

## IPH Storage Tank Heat Losses

The heat losses from the storage tanks are a combination of losses during daylight hours and losses during nighttime hours. The losses that occur during daylight hours differ from the losses that occur during nighttime hours due to the solar energy input to the collectors. At night, when no solar energy is present, the temperature of the airspace between the tank and the glazing is in equilibrium between the tank temperature and the temperature of the ambient air. However, the temperature of the airspace is not in equilibrium with the temperature of the tank and the ambient air temperature when solar energy is input to the collector. This condition causes a steeper thermal gradient to exist than the gradient present during nighttime hours and a greater heat loss coefficient then results.

## Daytime Heat Loss Coefficient

The heat loss coefficient that occurs during daylight hours was calculated from the stagnation curves shown in Figure 1 and Figure 2 and equations 1, 2 and 3. The stagnation curves show the change in temperature of the water in the storage tank as a function of time during heat-up. Heat input to the tank was calculated from equation 1.

1 $$Q_{in} = \frac{\Delta T}{\Delta t} * m_T$$

where

$Q_{in}$ = heat input to the tank (MJ/hr)

$\frac{\Delta T}{\Delta t}$ = temperature rise versus time (°C/hr)

$m_T$ = combined thermal mass of the tank and the water (MJ/°C)

The heat loss from the storage tank was computed from equation 2.

TABLE 1

| | |
|---|---|
| **Storage Tank Capacities:** | |
| IPH | 138.2 liters per tank |
| Batch | 183.6 liters |
| **Aperature Areas:** | |
| IPH | 2.245 $m^2$ per collector |
| Batch | 0.62 $m^2$ (est) |
| **Thermal Mass:** | |
| IPH (tank and water) | 588 KJ/°C per tank |
| Batch (tank & water) | 777 KJ/°C |
| Pipe (copper & water) | 1.55 KJ/°C/m |
| Pipe (PVC & water) | 1.31 KJ/°C/m |
| **Insulation Areas:** | |
| R7 rubber insulation | 0.339 $m^2$/m |
| R3 rubber insulation | 0.20 $m^2$/m |
| R7 foam insulation | 0.239 $m^2$/m |

## TABLE 1

| | |
|---|---|
| **Storage tank capacities** | |
| IPH | 138.2 l /tank |
| Batch | 183.6 l |
| **Aperature Areas** | |
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| **Thermal masses** | |
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| Batch (tank & water) | 777 KJ/°C |
| Pipe (copper & water) | 1.55 KJ/°C/m |
| Pipe (PVC & water) | 1.31 KJ/°C/m |
| **Insulation Areas** | |
| R7 rubber insulation | 0.239 $m^2$/m |
| R2 rubber insulation | 0.200 $m^2$/m |
| R7 foam insulation | 0.239 $m^2$/m |

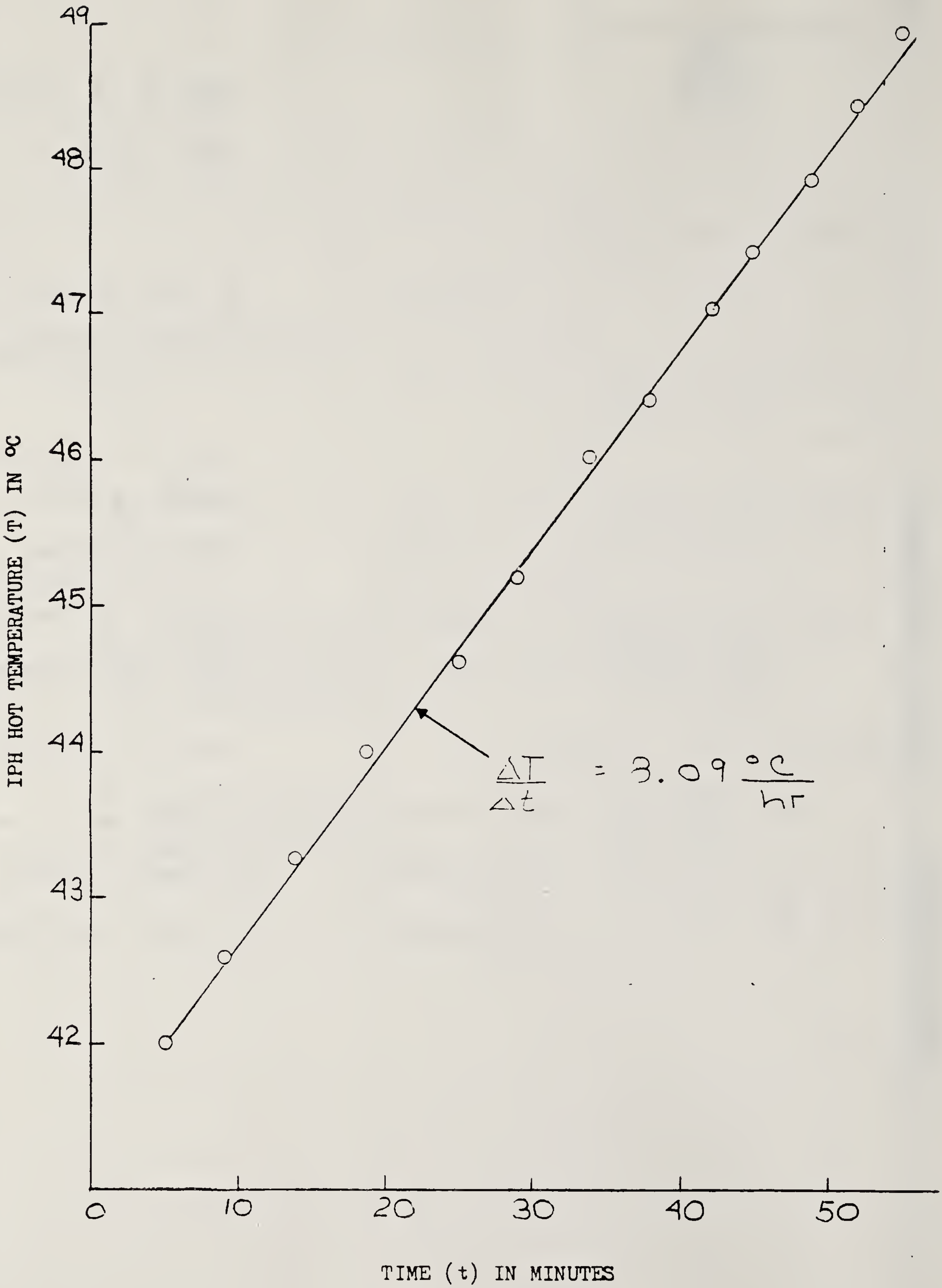


Figure 1: Stagnation heat-up curve for IPH hot tank for July 19

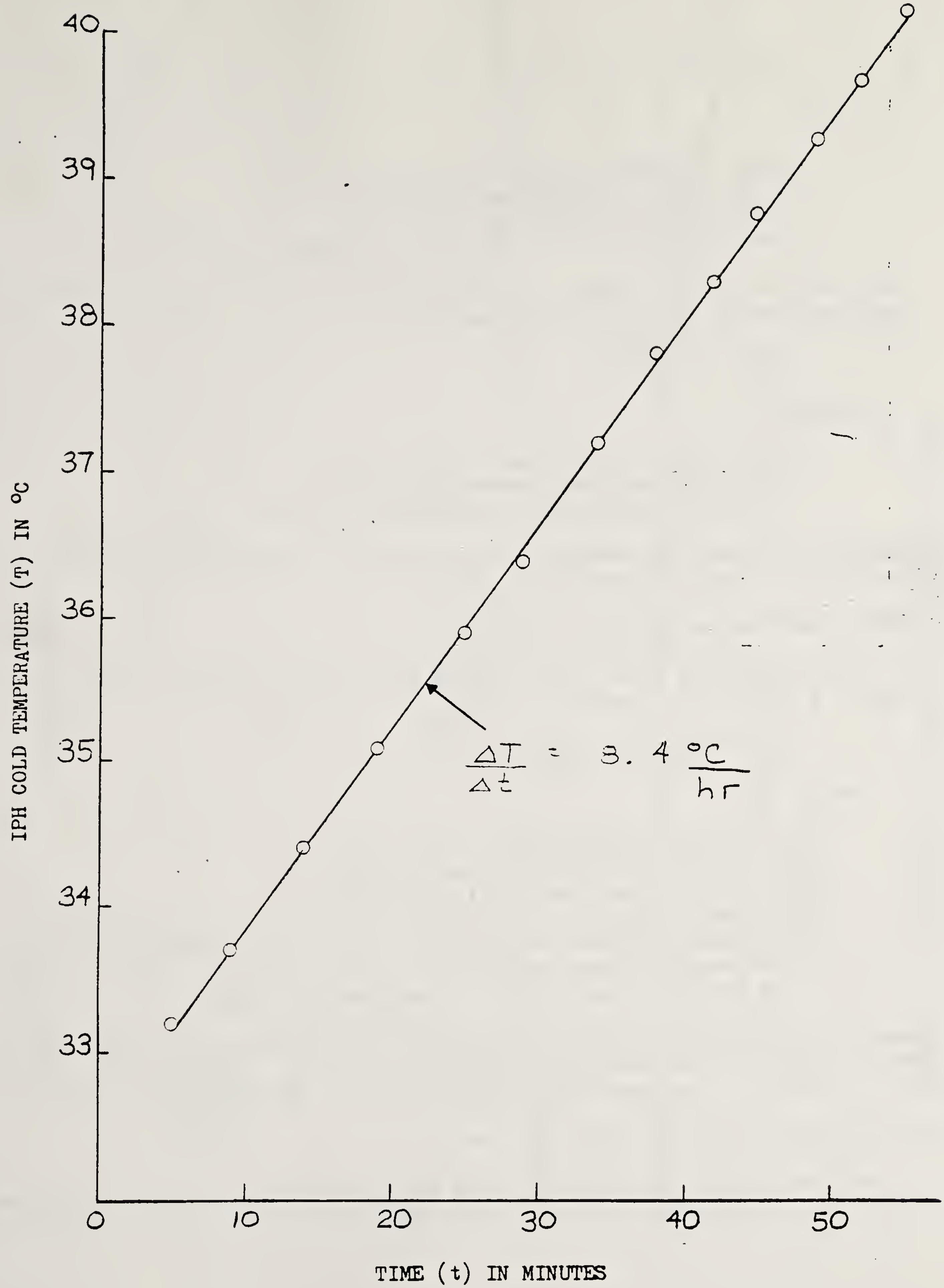


Figure 2: Stagnation heat-up curve for IPH cold tank for July 19

2 $$\text{Loss} = I * A * \text{trans} - Q_{in}$$

where

Loss = heat loss from the tank (MJ/hr)
I = solar insolation ($MJ/m^2$-hr)
A = area of collector ($m^2$)
trans = transmissivity of the glazing

The results of equations 1 and 2 yield the heat loss coefficient for daylight hours as shown in equation 3.

3 $$LC = \text{Loss}/(T_T - T_A)$$

where

LC = Loss coefficient (MJ/hr-°C)
$T_T$ = average tank temperature (°C)
$T_A$ = average ambient temperature (°C)

Appendix A shows a sample calculation for determining the heat loss coefficient for the hot tank.

Nighttime Heat Loss Coefficient

The nighttime heat loss coefficient was computed from the data of Figure 3 as shown in equation 4.

4 $$LC = \frac{T_1 - T_2}{\Delta t} * \frac{m_T}{T_T - T_A}$$

where

LC = Loss coefficient (MJ/hr -°C)
$T_1$ = initial tank temperature (°C)
$T_2$ = final tank temperature (°C)
$\Delta t$ = differential time element (hr)
$m_T$ = combined thermal mass of the tank and the water (MJ/°C)
$T_T$ = average tank temperature (°C)
$T_A$ = average ambient air temperature (°C)

Appendix A shows the calculations for determining the nighttime heat loss coefficient of the IPH cold tank. This value was found to be 15.0 KJ/hr-C and is within 5% of the SRCC value of 14.3 KJ/hr-°C.

Heat Loss of the IPH Pipe System

The heat loss of the IPH pipe system consisted of two components. These two components are heat losses during the drawdown and heat

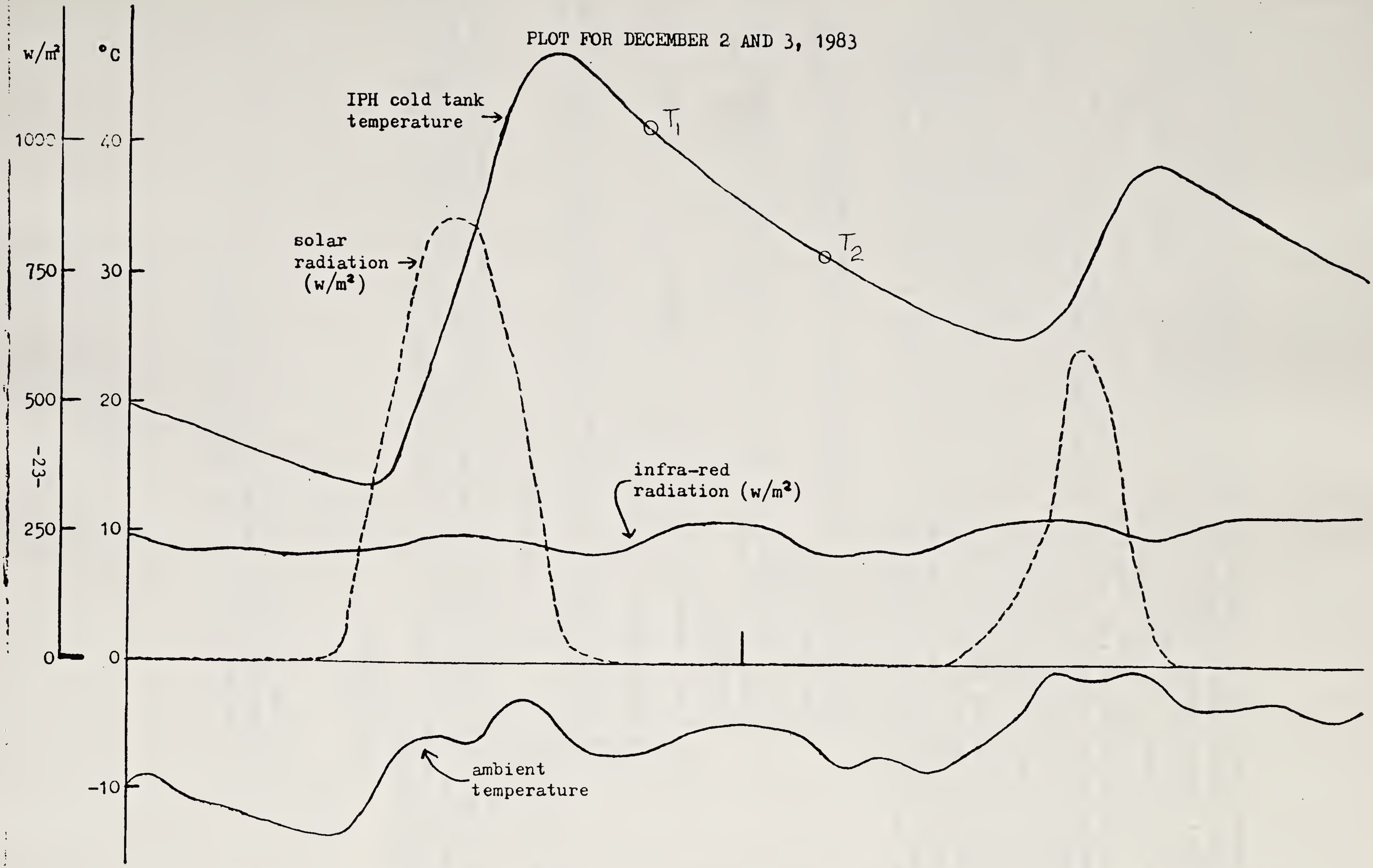


Figure 3: Stagnation cool-down curve for IPH cold tank

losses during the standby periods. An engineering estimate of the heat losses during the drawdown shows that these losses are less than 2% of the standby heat losses. Since the heat losses during drawdown are a small fraction of the heat losses during standby these losses are neglected in the analysis.

The standby heat losses of the IPH pipe system were assumed to behave as a Newtonian heating or cooling process. The process of Newtonian heating or cooling assumes that the material has negligible internal resistance to heat flow. Hence heat transfer is controlled solely by surface resistance.

The heat loss for the IPH pipe system was determined by summing the heat flows for each pipe section. Figure 4 shows the IPH pipe system. The pipe system consists of 20.0 mm ID copper pipe inside the house and 17.9 mm ID PVC pipe outside the house. The PVC pipe is insulated with R7 rubber insulation and the copper pipe is insulated with R3 rubber insulation. Calculation of the IPH pipe heat loss was based on the following assumptions:

1) Initial water temperature in the pipe downstream of the IPH hot tank is the average hot water temperature during the drawdown.

2) Initial temperature of the pipe upstream of the IPH cold tank is the average cold water temperature during the drawdown.

3) Pipe temperature and water temperature are the same.

4) Pipe has no resistance to heat flow.

5) Losses from the crossover pipe between the IPH cold tanks and the IPH hot tank are included with the IPH tank losses.

Calculation of the IPH Pipe Heat Flows

To calculate the heat flow for each section the temperature time constant and final water temperature were required. Equation 5 and equation 6 were used to calculate the time constant and final water temperature of each section respectively. The heat flow of each section was computed from equation 7.

5 $$T = e^{-\left(\frac{UA}{mr}\right)\theta}$$

where

T = time constant
U = thermal conductance of the insulation (KJ/hr-m -°C)
A = surface area of the insulation ($m^2/m$)

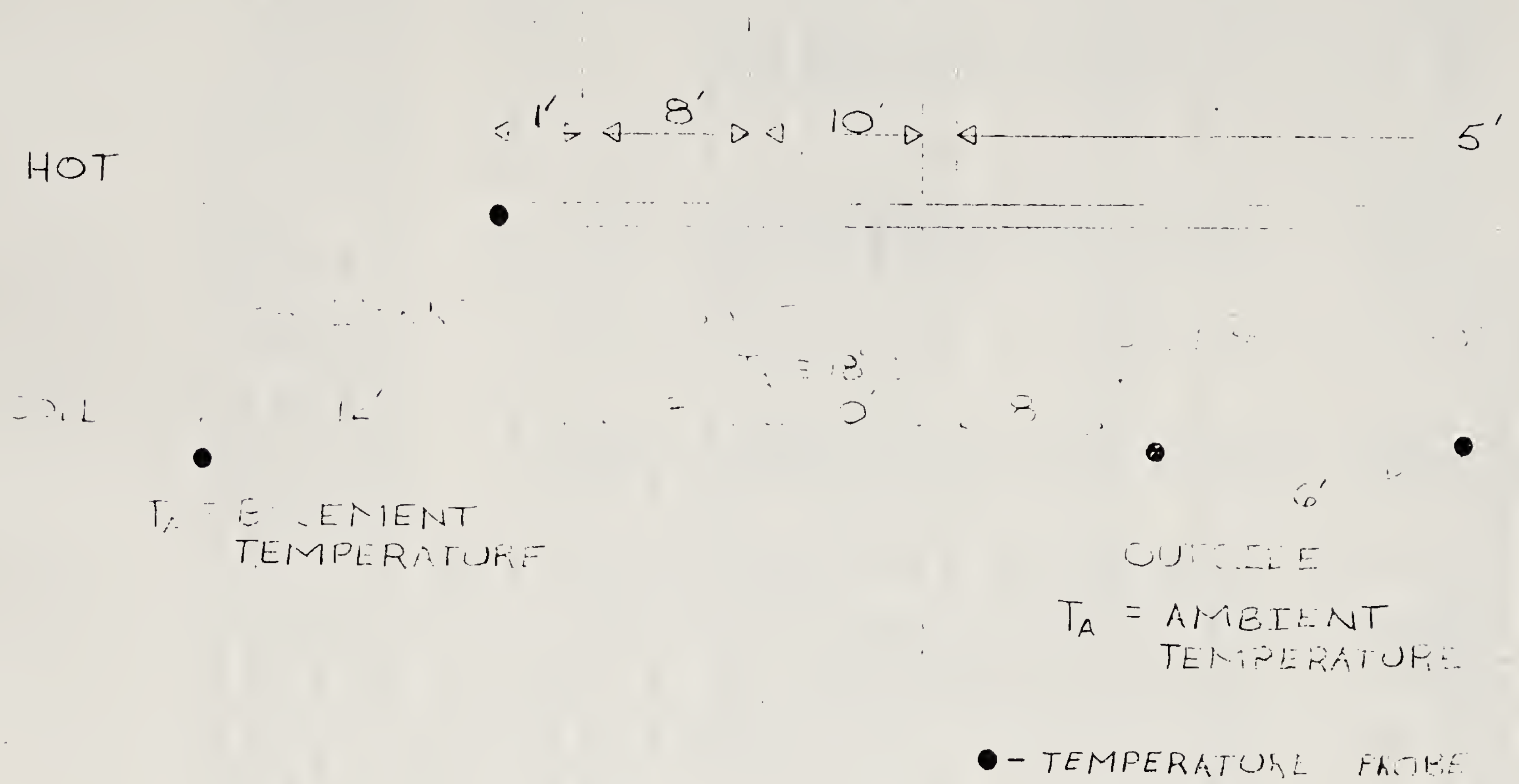


Figure 4:  IPH heat system

$m_T$ = combined thermal mass of the pipe and the water (KJ/°C/m)
$\theta$ = time (hr)

6 $$T_2 = (T_1 - T_A) * T + T_A$$

where

$T_2$ = final pipe temperature ( C)
$T_1$ = initial pipe temperature ( C)
$T_A$ = local air temperature (basement, house or ambient) (°C)
$T$ = time constant

7 $$Q = (T_2 - T_1) * m_T * l$$

where

Q = heat flow (KJ)
$m_T$ = combined thermal mass of the pipe and the water (KJ/°C/m)
$l$ = pipe length (m)

Shown in Appendix A is a sample calculation for the heat flow for one section of the IPH pipe system.

Summary of the IPH Heat System

Shown in Table 2 are the monthly summaries for the IPH heat system. During the first 12 months of operation the system delivered 49% of the heat required by the load; the load being defined as the heat delivered by the electric hot water heater at 50 C delivery temperature. The system used 35% of the solar energy striking the collectors. Solar energy is defined as the solar insolation striking the collector multiplied by the collector area. Also during this time the combined IPH tank losses were 67% of the heat delivered to the load.

Batch Solar Heat System Heat Losses

The batch heat system heat losses consisted of the same heat loss components that affected the IPH heat system. That is, heat losses from the storage tank and heat losses from the pipe network.

Batch Storage Tank Heat Losses

As with the IPH system the heat loss of the batch storage tank was a combination of losses that occurred during daylight hours and losses that occurred during nighttime hours. The daytime heat loss coefficient differed from the nighttime heat loss coefficient due to the temperature of the airspace not being in equilibrium between the temperature of the tank and the ambient air temperature.

TABLE 2

Monthly Summaries of IPH Heat System

| MONTH | AMBIENT TEMP (°C) | SOLAR INSOLATION (MJ) | IPH HEAT (MJ) | ELECTRIC HEAT (MJ) | TANK LOSS (MJ) | PIPE LOSS (MJ) |
|---|---|---|---|---|---|---|
| NOVEMBER | -2.7 | 805 | 185 | 1455 | 248 | 49 |
| DECEMBER | -11.8 | 1025 | 65 | 1665 | 464 | 14 |
| JANUARY | -3.40 | 1200 | 230 | 1700 | 367 | 33 |
| FEBRUARY | -1.65 | 1890 | 690 | 1575 | 518 | 25 |
| MARCH | 2.25 | 1945 | 795 | 1630 | 489 | 23 |
| APRIL | 4.60 | 2330 | 900 | 1565 | 557 | 2.7 |
| MAY | 9.77 | 2350 | 920 | 1490 | 461 | -13 |
| JUNE | 13.9 | 2595 | 990 | 1255 | 461 | -39 |
| JULY | 19.5 | 2695 | 1070 | 1225 | 618 | -55 |
| AUGUST | 18.8 | 2945 | 1170 | 1220 | 530 | -75 |
| SEPTEMBER | 10.3 | 2410 | 830 | 1235 | 550 | -52 |
| OCTOBER | 4.17 | 2090 | 700 | 1440 | 497 | -14 |
| | | | | | | |
| TOTAL | | $2.42 \times 10^4$ | 8545 | $1.75 \times 10^4$ | 5760 | -101 |

### Daytime Heat Loss Coefficient

The daytime heat loss coefficient is calculated from Figure 5 and equations 1, 2 and 3. However, these equations yielded an erroneous result. That result being a negative heat loss coefficient. The only possible explanation for this value is that the water in the storage tank is stratifying during the standby period and that the temperature probe in the tank does not indicate the average temperature of the water.

### Nighttime Heat Loss Coefficient

The nighttime heat loss coefficient was calculated from equation 4 which yielded a value of 22.0 KJ/hr-°C. This value is 47% larger than the nighttime heat loss coefficient for the IPH storage tanks. The larger value is due to a larger thermal mass and less insulation around the tank.

### Batch Pipe System

The batch pipe system is shown in Figure 6 and consists of 20 mm ID copper tubing. R7 foam insulation is used to insulate the portion of the pipe that is outside of the house and R3 rubber insulation is used to insulate the pipe on the inside of the house.

### Batch Pipe Heat Losses

As with the IPH pipe system the batch pipe system has heat losses consisting of two components; these two components are heat losses that occur during the drawdown and heat losses that occur during standby periods. An engineering analysis of the losses during the drawdown showed that these losses are 1.4% of the losses during the standby periods. Therefore these losses were neglected in the analysis.

Computation of the batch pipe system heat losses was performed in the same manner as the IPH pipe losses. A Newtonian cooling or heating process was assumed to occur and the following assumptions were also made:

1) Initial water temperature in the pipe downstream of the batch storage tank is the average hot water temperature during the draw.

2) Initial temperature of the water upstream of the batch tank is the average cold water temperature during the draw.

3) Pipe temperature and water temperature are the same.

4) Pipe has no resistance to heat flow.

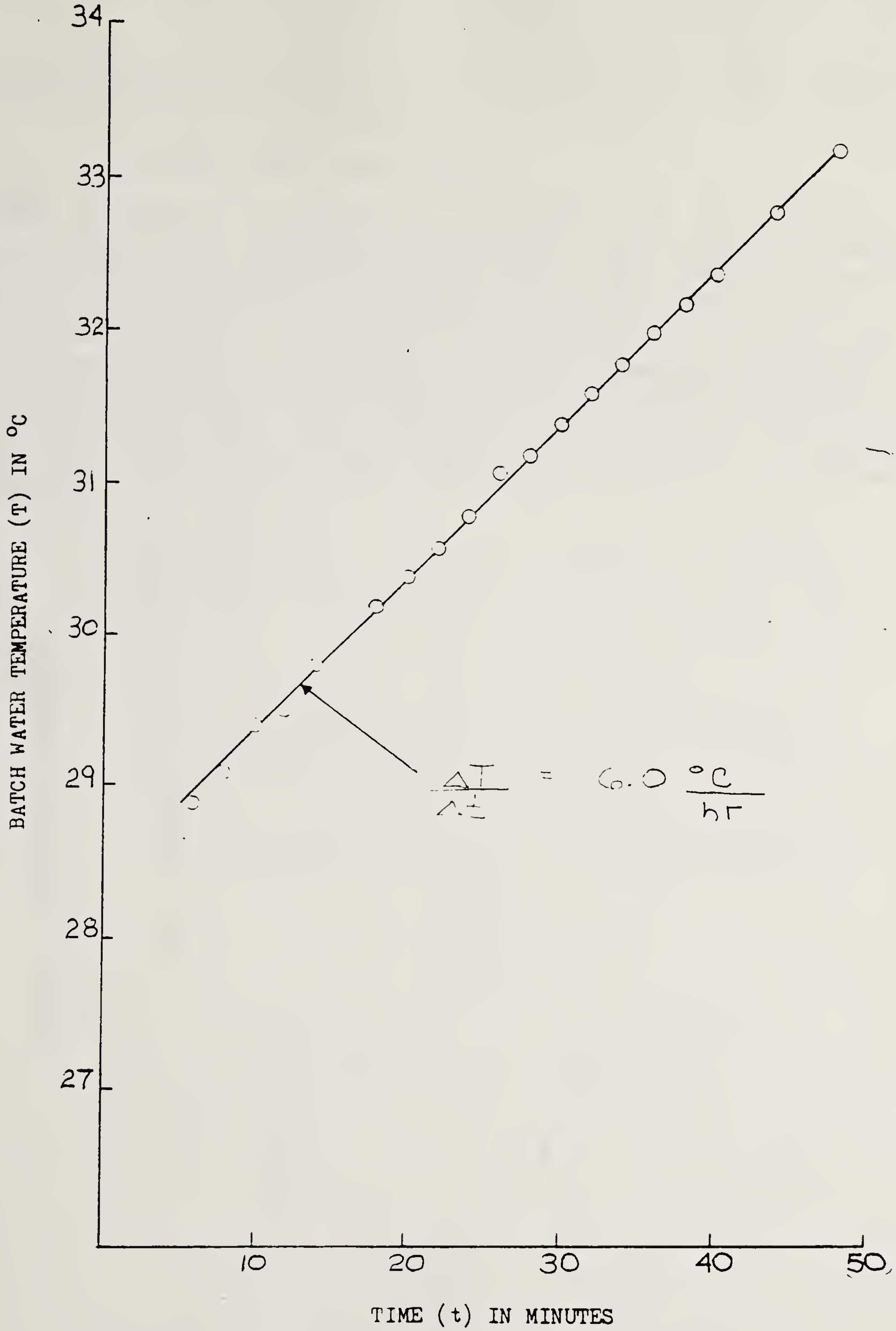


Figure 5: Stagnation heat-up curve for Batch storage tank

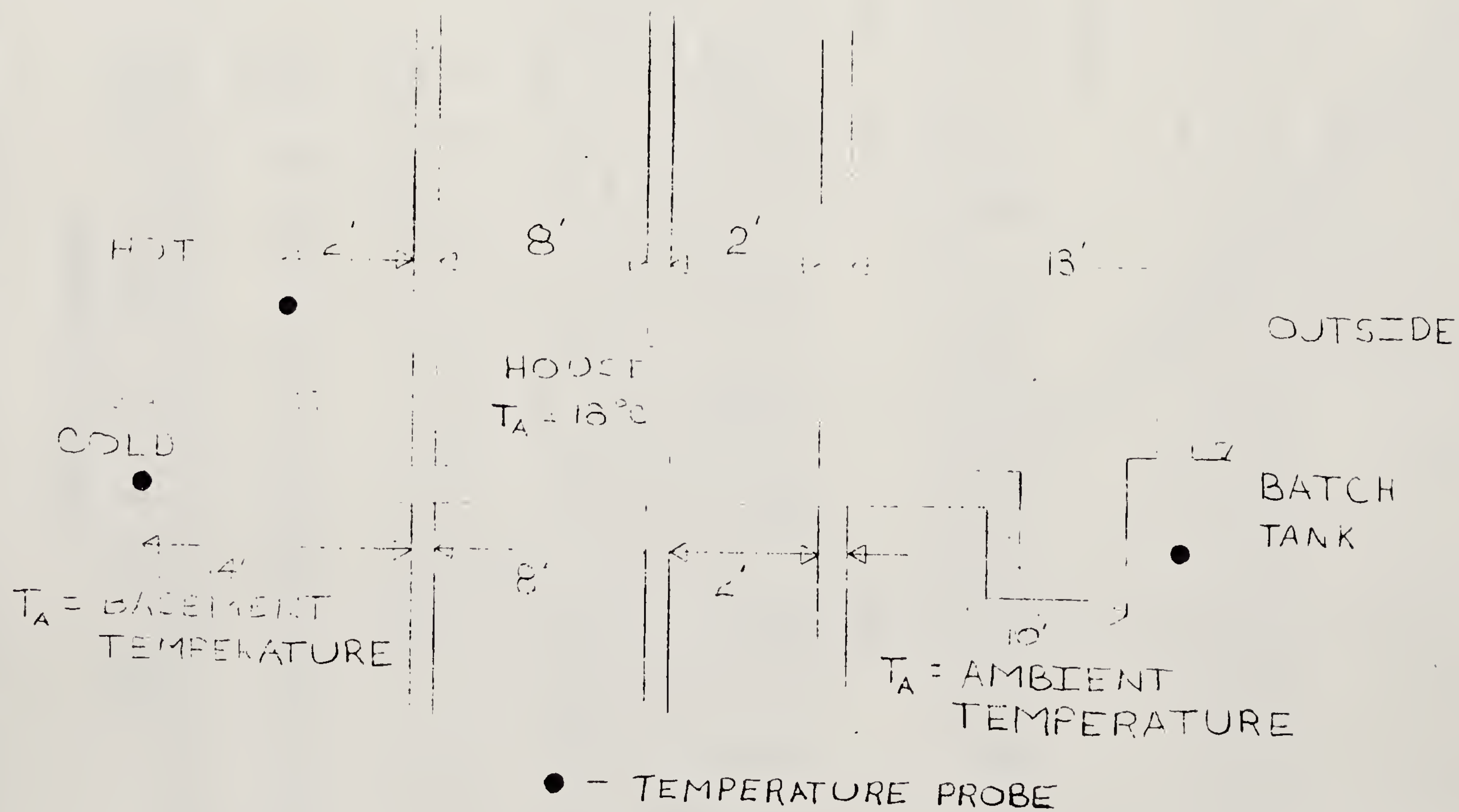


Figure 6: Batch heat system

Equations 5, 6 and 7 are then used to calculate the heat flow for each section of the batch pipe system.

## Summary of the Batch Solar Heat System

Shown in Table 3 are the monthly summaries for the batch heat system. During the 7 months that the batch heat system was operated it delivered 20% of the heat required by the load; again the load is defined as the heat generated by the electric hot water heater at 50 C delivery temperature. The batch system used 76% of the solar energy striking the collector. This high efficiency is due to the lower operating temperature of the batch system and hence the batch system has smaller heat losses to the environment. The batch pipe losses were actually heat gains into the pipe network and this heat gain represents 9% of the heat delivered to the load. The heat gain into the batch pipe network is due to the lower operating temperature of the batch heat system.

TABLE 3

Monthly Summaries of Batch Heat System

| MONTH | AMBIENT TEMP (°C) | SOLAR INSOLATION (MJ) | BATCH HEAT (MJ) | ELECTRIC HEAT (MJ) | PIPE LOSS (MJ) |
|---|---|---|---|---|---|
| | | | | | |
| APRIL | 4.60 | 320 | 245 | 1565 | 44 |
| MAY | 9.77 | 325 | 260 | 1490 | 39 |
| JUNE | 13.9 | 360 | 290 | 1255 | 23 |
| JULY | 19.5 | 370 | 340 | 1225 | 16 |
| AUGUST | 18.8 | 405 | 340 | 1220 | 13 |
| SEPTEMBER | 10.3 | 330 | 205 | 1235 | 6 |
| OCTOBER | 4.17 | 290 | 155 | 1440 | 22 |
| | | | | | |
| TOTAL | | 2400 | 1835 | 9340 | 163 |

APPENDIX A

$$Q_{in} = \frac{\Delta T}{\Delta t} \cdot \Delta t \cdot m_{Thrm}$$

$$= 8.09 \ \frac{°C}{hr} \cdot 0.588 \ MJ/°C$$

$$= 4.76 \ MJ/hr$$

$$\text{"LOSS"} = I \cdot A \cdot trans - Q_{in}$$

$$= 3.39 \cdot 2.245 \cdot 0.7 - 4.76$$

$$= 0.56 \ MJ/hr$$

$$\text{Loss Coef} = \frac{\text{"LOSS"}}{T_{TANK} - T_{AMB}}$$

$$= \frac{0.56 \ MJ/hr}{(45.4 - 26.2)°C}$$

$$= 0.0294 \ \frac{MJ}{hr\text{-}°C}$$

Sample calculation for the heat loss coefficient of the IPH hot tank

Data for July 19

$$T_1 = 38.47\,°C$$

$$T_2 = 33.21\,°C$$

$$T_T = 35.74\,°C$$

$$T_A = -5.48\,°C$$

$$\Delta t = 5\ hr$$

$$m_T = 0.588\ MJ/°C$$

$$LC = \frac{T_1 - T_2}{\Delta t} \cdot \frac{m_T}{T_T - T_A}$$

$$= \frac{(38.47 - 33.21)\,°C}{5\ hr} \cdot \frac{0.588\ MJ/°C}{(35.74 - (-5.48))\,°C}$$

$$= 0.0150\ \frac{MJ}{hr\ °C}$$

Sample calculation for the nighttime heat loss coefficient of the IPH cold storage tank

Data for December 2 - December 3

$$Q = U \cdot A \cdot \Delta T \cdot L \cdot t$$

Outside hot:

$$Q = \frac{1}{7} \frac{BTU}{hr \cdot ft^2 \cdot {}^\circ F} \cdot (0.339)\frac{m^2}{m} \cdot (10-30)^\circ C \cdot (1.524)m \cdot \left(\frac{22}{60}\right)hr$$

$$\cdot (1.8)\frac{{}^\circ F}{{}^\circ C} \cdot 1.055\,\frac{KJ}{BTU} \cdot \frac{10.76\ ft^2}{m^2}$$

$$= -11.07\ KJ$$

Outside cold:

$$Q = \frac{1}{7}(0.339)(10-6)^\circ C \cdot (2.438) \cdot \left(\frac{22}{60}\right)(1.8)(1.055)(10.76)$$

$$= 3.543\ KJ$$

Inside hot:

$$Q = \frac{1}{3}(0.200)(18-30)^\circ C \cdot (5.791)\left(\frac{22}{60}\right)(1.8)(1.055)(10.76)$$

$$= -34.66\ KJ$$

Inside cold

$$Q = \frac{1}{3}(0.200)(18-6)^\circ C \cdot (9.144)\left(\frac{22}{60}\right)(1.8)(1.055)(10.76)$$

$$= 54.73\ KJ$$

$$\therefore \bar{Q}_{TOTAL} = 12.54\ KJ/day$$

$$|\bar{Q}|_{\substack{pipe \\ stagnation}} = 985\ KJ/day$$

$$\frac{\bar{Q}_T}{\bar{Q}_{PS}} = 0.013 \quad \text{or} \quad 1.3\%$$

Sample calculation for the heat loss of the IPH pipe system during the during the draw versus the heat loss during the standby periods

$U = 2.919 \ \mathrm{KJ/hr\text{-}m^2\text{-}^\circ C}$ $\qquad$ $T_A = 20.4\,^\circ\mathrm{C}$

$A = 0.339 \ \mathrm{m^2/m}$ $\qquad$ $T_1 = 34.3\,^\circ\mathrm{C}$

$m_T = 1.31 \ \mathrm{KJ/^\circ C\text{-}m}$ $\qquad$ $L = 1.524 \ \mathrm{m}$

$$\begin{aligned} T &= e^{-\left(\frac{UA}{m_T}\right)\theta} \\ &= e^{-\left(\frac{2.919 \cdot 0.339}{1.31}\right)\cdot 1hr} \\ &= 0.471 \end{aligned}$$

$$\begin{aligned} T_2 &= (T_1 - T_A) \cdot T + T_A \\ &= (34.3 - 20.4) \cdot 0.471 + 20.4 \\ &= 26.9\,^\circ\mathrm{C} \end{aligned}$$

$$\begin{aligned} Q &= (T_2 - T_1) \cdot m_T \cdot L \\ &= (26.9 - 34.3) \cdot 1.31 \cdot 1.524 \\ &= -14.7 \ \mathrm{KJ} \end{aligned}$$

Sample calculation for one section of the IPH pipe system

Data for 11:00 AM, July 24

APPENDIX B

$T_1 = 20.8\,°C$

$T_2 = 20.35\,°C$

$T_A = 10.0\,°C$

$T_T = 20.6\,°C$

$m_T = 0.777\ MJ/°C$

$\Delta t = 1.5\ hr$

$$UA = \frac{T_1 - T_2}{\Delta t} \cdot \frac{m_T}{T_T - T_A}$$

$$UA = \frac{20.8 - 20.35}{1.5} \cdot \frac{0.777}{20.6 - 10.0}$$

$$= 0.022\ \frac{MJ}{hr \cdot °C}$$

Sample calculation for the nighttime heat loss coefficient of the batch storage tank

Data for July 1

$Q = U \cdot A \cdot \Delta T \cdot L \cdot t$

Outside hot:

$$Q = \frac{1}{7} \frac{BTU}{hr \cdot ft^2 \cdot °F} \cdot 0.785 \frac{ft^2}{ft} \cdot (10-25)°C \cdot 13ft \cdot \left(\frac{22}{60}\right) hr$$

$$\cdot 1.8 °F/°C \cdot 1.055 \frac{KJ}{BTU} = 15.2 \ KJ$$

Outside cold:

$$Q = \frac{1}{7}(0.785)(10-6)(10)\left(\frac{22}{60}\right)(1.8)(1.055)$$

$$= 3.12 \ KJ$$

Inside hot:

$$Q = \frac{1}{3}(0.655)(18-25)(12)\left(\frac{22}{60}\right)(1.8)(1.055)$$

$$= -12.8 \ KJ$$

Inside cold:

$$Q = \frac{1}{3}(0.655)(18-6)(24)\left(\frac{22}{60}\right)(1.8)(1.055)$$

$$= 43.8 \ KJ$$

$$\bar{Q}_{TOTAL} = 18.9 \ \frac{KJ}{day}$$

$$\bar{Q}_{\substack{pipe \\ stagn.}} = 1326 \ \frac{KJ}{day}$$

$$\frac{\bar{Q}_T}{\bar{Q}_{P.S}} = 0.014 \quad \text{or} \quad 1.4\%$$

Sample calculation for the heat loss of the Batch pipe system during the draw versus the heat loss during the standby period

# MONTANA POWER COMPANY
## SOLAR WATER HEATER PERFORMANCE ANALYSIS

LIQUID COLLECTOR MODEL SOLARCRAFT
STATE INDUSTRIES
Frta= .744  FrUl= .775 Btu/F-sft
Tank cap.= 80 gal.,  R value = 14
Tilt= 45 deg   Area= 46.5 sft
Heat exchanger? YES

CHARLESS FOWLKES
FOWLKES ENGINEERING
31 GARDNER PARK DRIV
BOZEMAN, MT 59715
406-587-3779
9/18/84

### f-CHART PERFORMANCE FOR BOZEMAN MT.

| MON | SOLAR RAD Btu/sft | AMBIENT TEMP degF | HOT WATER LOAD Btu | SOLAR FRACT % | SOLAR ENERGY Btu | AUXILIAR ENERG Bt |
|---|---|---|---|---|---|---|
| JAN | 26,700 | 23.4 | 1,376,990 | 34 | 474,171 | 902,81 |
| FEB | 32,760 | 25.9 | 1,376,990 | 44 | 619,610 | 757,37 |
| MAR | 45,2[illegible]0 | 30.2 | 1,376,990 | 63 | 876,205 | 500,78 |
| APR | 50,1[illegible]0 | 42.1 | 1,376,990 | 71 | 979,102 | 397,88 |
| MAY | 44,450 | 50.9 | 1,376,990 | 64 | 894,741 | 482,24 |
| JUN | 52,9[illegible]0 | 57.6 | 1,376,990 | 75 | 1,045,958 | 331,03 |
| JUL | 56,100 | 66.3 | 1,376,990 | 80 | 1,109,271 | 267,71 |
| AUG | 52,350 | 64.9 | 1,376,990 | 75 | 1,045,863 | 331,12 |
| SEP | 49,[illegible]80 | 55.6 | 1,376,990 | 71 | 983,022 | 393,96 |
| OCT | 44,570 | 46.1 | 1,376,990 | 63 | 879,738 | 497,25 |
| NOV | 26,[illegible]00 | 32.[illegible] | 1,376,990 | 35 | 489,005 | 887,98 |
| DEC | 19,950 | 25.[illegible] | 1,376,990 | 22 | 314,527 | 1,062,46 |
| YEAR | 501,[illegible] | 43.1 | 16,523,881 | 58 | 9,711,218 | 6,812,66 |

Average annual system efficiency is 41%

### ECONOMIC ANALYSIS OF SYSTEM

| | |
|---|---|
| System cost.............. | $3,600 |
| Fed. tax credit......... | $1,440 |
| State tax credit........ | $115 |
| Net system cost......... | $2,045 |
| NET COST/MMBtu/yr. | $210.58 |
| NET COST/MMBtu 20yr. | $10.52 |

Current electric cost is $8.51/MMBtu
Current gas cost is $5.25/MMBtu

(1) Assume 4%/year real escalation
(2) Assume 6%/year real escalation
Assume 65% burner efficiency

ACCUMULATED SOLAR SAVINGS FROM JAN. 1983

| YEAR ENDING | ELECTRIC(1) | GAS(2) |
|---|---|---|
| 1983 | $83 | $78 |
| 1984 | $165 | $157 |
| 1985 | $251 | $240 |
| 1986 | $341 | $328 |
| 1987 | $434 | $422 |
| 1988 | $530 | $521 |
| 1989 | $631 | $626 |
| 1990 | $735 | $737 |
| 1991 | $844 | $855 |
| 1992 | $957 | $980 |
| 1993 | $1,075 | $1,112 |
| 1994 | $1,197 | $1,253 |
| 1995 | $1,324 | $1,402 |
| 1996 | $1,457 | $1,559 |
| 1997 | $1,594 | $1,727 |
| 1998 | $1,737 | $1,904 |
| 1999 | $1,886 | $2,092 |
| 2000 | $2,041 | $2,291 |

f-chart version MFC1

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